Dynamic Bellows for a Pulse Tube Cryocooler Application

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Electrified aircraft propulsion systems will require powertrain components that are both lightweight and highly efficient. NASA's 1.4 MW High Efficiency Megawatt Motor (HEMM) has been designed to run at 6800 RPM with >98% efficiency and 16 kW/kg, achieving low losses by incorporating a cryocooler within its rotor shaft to cool superconducting DC windings. A pulse-tube cryocooler typically contains an oscillating piston that drives acoustic waves; however, leakage around the piston reduces the efficiency of the cryocooler. A metal bellows was designed to replace the piston to allow for little to no leakage. The requirements for this bellows were extreme in terms of displacement amplitude, oscillating pressure amplitude, temperature, and frequency. It was determined that an off-the-shelf edge-welded bellows would not work for this application, so a custom bellows was designed to include full convolutions welded only on their outer diameters where the stress is lower. Several high-fatigue-strength materials were considered, including 17-4PH H900 and AerMet 100. The final bellows design includes 74 convolutions with a variable inner diameter that helps distribute and reduce the dynamic stress throughout the length rather than concentrating the high stress in one location. To test the concept, a short 10-convolution bellows with constant inner diameter was designed for testing. One short bellows test article was fabricated with the outer diameter weld, and one by 3D printing. Both will be fatigue-tested with the goal of achieving 10^7 cycles.

I. Introduction

NASA has been investigating electrified aircraft propulsion as a means of improving efficiency and reducing fuel burn in commercial aircraft. Achieving these improvements requires electric components with very high efficiency and power density. Requirements for the High Efficiency Megawatt Motor (HEMM) are based on the NASA Single-Aisle Turboelectric Aircraft – Aft Boundary Layer (STARC-ABL) concept aircraft [1] to produce 1.4 MW of power with >98% efficiency and 16 kW per kg of electromagnetic mass [2, 3, 4]. This efficiency is accomplished by utilizing a wound field rotor with superconducting windings resulting in very low losses. These windings also generate a very large rotating magnetic field (over 1T) through the air gap to the slotless stator windings, allowing for a high power density. The superconducting windings are cooled by a pulse-tube cryocooler embedded within the rotor.

A cross section of the HEMM motor is shown in Fig. 1, detailing the components. The rotor incorporates a 2.5 kW pulse-tube cryocooler (shown in blue in Fig. 1) within the rotor shaft, designed to lift 50 W to produce a 50 K temperature at the cold tip, conductively cooling the superconducting rotor windings [5]. The acoustic pulses are created within high pressure helium through a piston that is driven by a linear motor (shown in green in Fig. 1).

The cryocooler needs to have low losses to keep the overall motor efficiency high. Traditional pulse tube cryocooler designs use a linear motor that drives the piston to pump the gas. The piston is supported by multiple flexures, or flat spiral shaped springs, that function as a bearing as well as a spring return. Leakage around the piston was identified as contributing to losses, decreasing the predicted lift by more than 10%. A bellows was proposed as a replacement for that piston. The piston is shown in Fig. 2, and the bellows concept is shown in Fig. 3. The original idea was that the bellows would occupy a similar space as the piston, with the same effective diameter and stroke.

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The requirements for the bellows are given in Table 1, though the axial length needed to be increased from the original value to keep stresses low. The bellows displacement and differential pressure, which are 30 degrees out of phase, are shown in Fig. 4. The phasing between the two produce optimal conditions for heat lift within the pulse tube cryocooler.

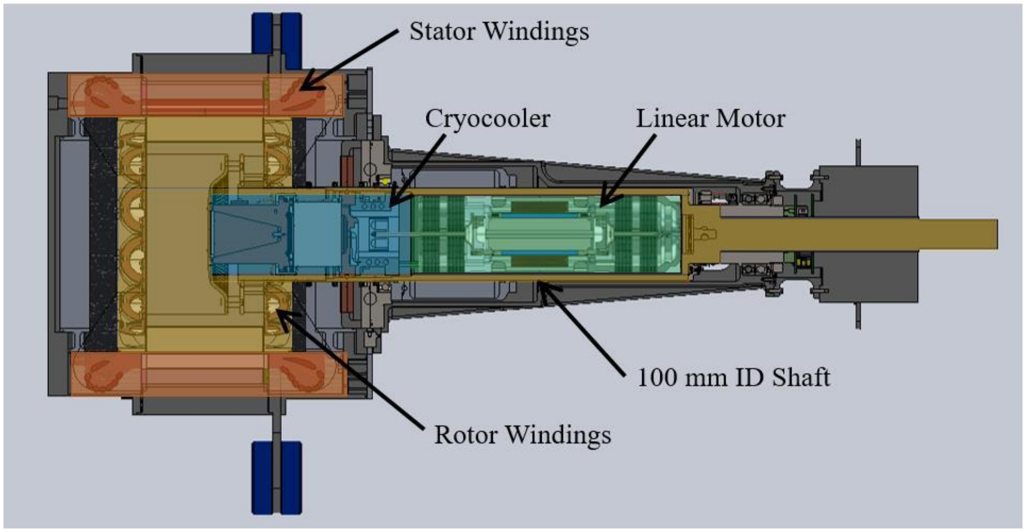


Fig. 1 HEMM motor assembly

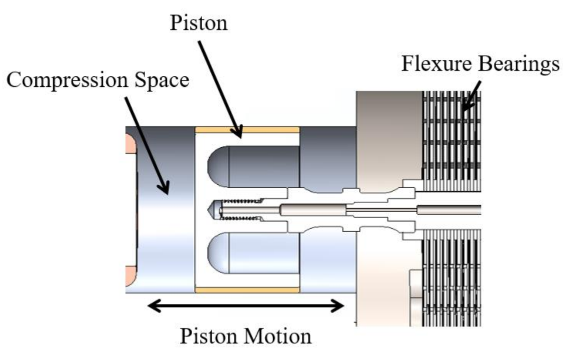


Fig. 2 Piston concept

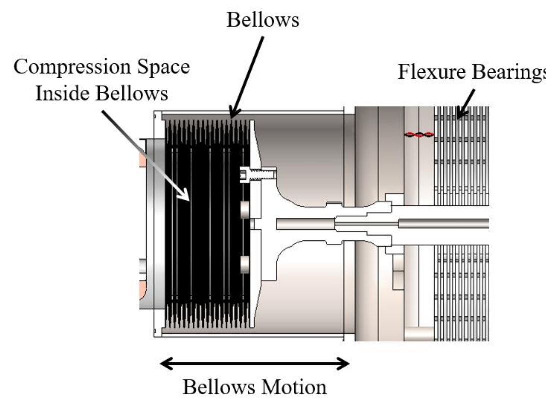


Fig. 3 Bellows concept

Table 1 Bellows requirements

Parameter	**Value**
Temperature	Up to 200°C (392°F)
Environment	Helium
Nominal Pressure	6.27 MPa (910 psi)
Pressure Amplitude	0.828 MPa (120 psi)
Displacement Amplitude	± 13 mm (0.51 in)
Piston Diameter	44.3 mm (1.74 in)
Axial Length (preferred)	60.0 mm (2.36 in)
Frequency	55-60 Hz
Cycles	> 10^7

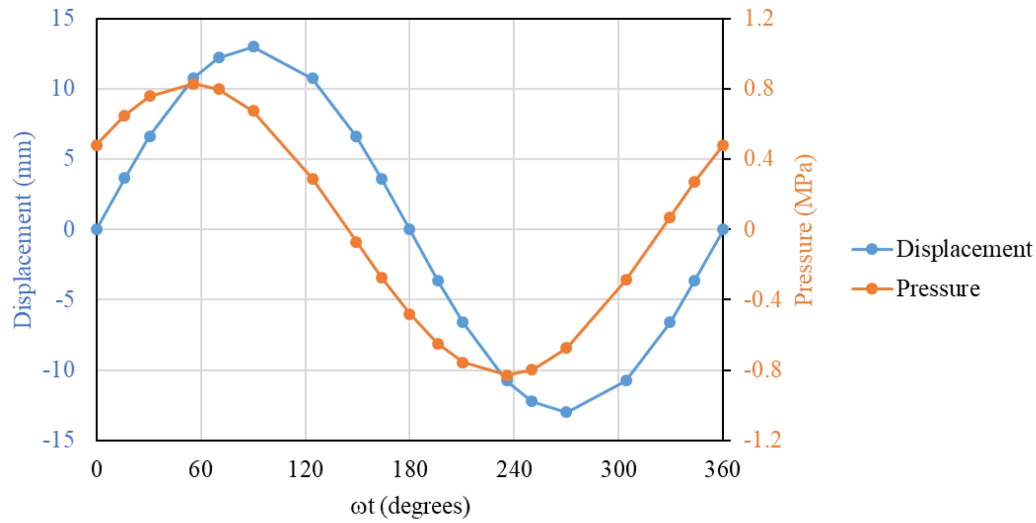


Fig. 4 Displacement of and pressure differential across bellows (ω is bellows angular frequency, t is time)

II. Bellows Geometry

The bellows is required to withstand a substantial axial displacement while subject to an oscillating pressure differential, all at near 60 Hz. Henschel et al. [6] describe many different bellows plate (washer) configurations, including flat plate (e.g. Fig. 5a and 5b), corrugated shapes (e.g. Fig. 5c and 5d), and toroidal and U-shaped (not shown). Each shape has its advantages and disadvantages. Toroidal and U-shapes do not have capability for a longer stroke, so were not considered for this application. The flat plate shapes have higher axial stiffness than the corrugated shapes, and both flat and swept shapes have good resistance to differential pressure. The corrugated shapes do require fabrication with a weld at the inner diameter.

The maximum stress location will be in the slot at the inner diameter, and for a given axial displacement amplitude, that maximum stress will be higher for a shorter bellows. Due to axial space constraints, a solid piece construction at the inner diameter is preferred (rather than welded or brazed for example) to maximize strength at that location. This means that the bellows cannot be made of individual washers welded at both the inner and outer diameters. Rather, the bellows will be made as a solid piece by 3D printing, or by welding together at the outer diameter individual disks with machined outer slots (Fig. 6). A convolution is basically defined as two individual washers connected at their inner diameter with a slot between. To start, the flat shape (Fig. 5a) was chosen for the test articles, though there is capability of having a shaped inner slot (Fig. 5e).

a. Flat b. Flat conical c. Swept d. Nested ripple e. Flat outer slot

Fig. 5 Bellows convolution shapes

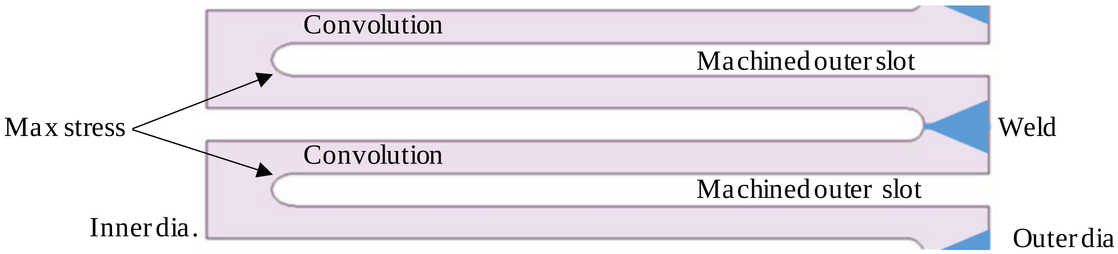
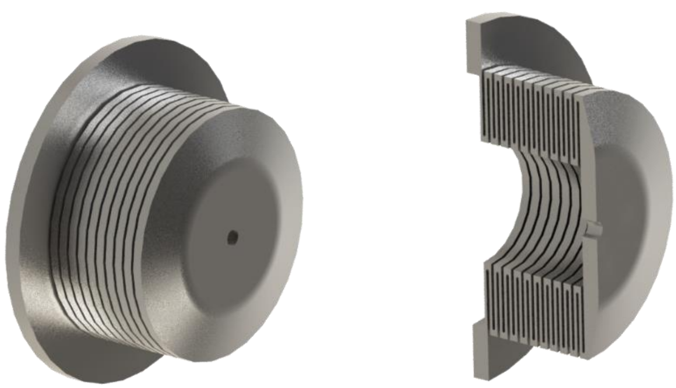


Fig. 6 Bellows with convolutions welded at outer diameter

The basic geometry of the bellows is shown in Fig. 7 for a short 10-convolution bellows with a flange at one end and an end cap at the other. There are bellows available for purchase off-the-shelf, but none were recommended for this type of application. For example, edge-welded bellows with the nested ripple design were not recommended by the manufacturer for large numbers of cycles, or for oscillating pressure (vs. positive pressure differential).

With regard to replacing a piston with a bellows in a cryocooler application, there are a couple of specific points to note. First, the piston is part of a defined mass-spring-damper system that includes the linear motor moving mass as well as the flexure bearing stiffness and moving mass. It also couples to the acoustic assembly of the cryocooler. This mass-spring-damper system is designed to optimize the cryocooler performance. The bellows will have a different moving mass than the piston, and will also add stiffness. Second, the bellows will introduce additional gas volumes within the convolutions on either side of the bellows. These volumes will also need to be accounted for in the cryocooler design and optimization.



a. Short bellows **b. Cross section**
Fig. 7 Bellows with 10 convolutions and end fixtures

III. Simple Stress and Frequency Analysis

It was clear in early studies that a bellows that could meet the displacement and pressure requirements would require many convolutions, and be much longer than the original length constraint. This means that the fundamental axial resonance frequency can be quite low, resulting in dynamic stresses much higher than the static stress induced by the displacement and pressure. Therefore minimizing dynamic stress means maximizing the fundamental axial resonance frequency. A very simple analysis of stress and resonance shows the essential material properties for optimization.

Consider a flat washer shape that is fixed at its inner diameter and displaced at its outer diameter (Fig. 8), and assume the ratio of radius to span *s* is large. The stress in one washer is defined by

$$\sigma_1 \cong \frac{\Delta P}{2}\left(\frac{s}{t}\right)^2 + \left(\frac{3E}{1-\nu^2}\right)\left(\frac{t}{s^2}\right)(\delta) \quad (1)$$

where t is the washer thickness, the span is $s = R_o - R_i$, R_o and R_i are the outer and inner diameters, respectively, ΔP is the pressure difference across the washer, δ is the axial deflection, E is Young's modulus, and ν is Poisson's ratio [6]. Looking at the stress equation, we can see that a shorter span s will minimize stress due to pressure difference ΔP, and a longer span s will minimize stress due to deflection δ. If there is a stack of N washers that are deflected by δ, then the stress is

$$\sigma_N \cong \frac{\Delta P}{2}\left(\frac{s}{t}\right)^2 + \left(\frac{3E}{1-\nu^2}\right)\left(\frac{t}{s^2}\right)\left(\frac{\delta}{N}\right) \tag{2}$$

Clearly some optimization of s, t, and N will be required to minimize stress for both ΔP and δ.

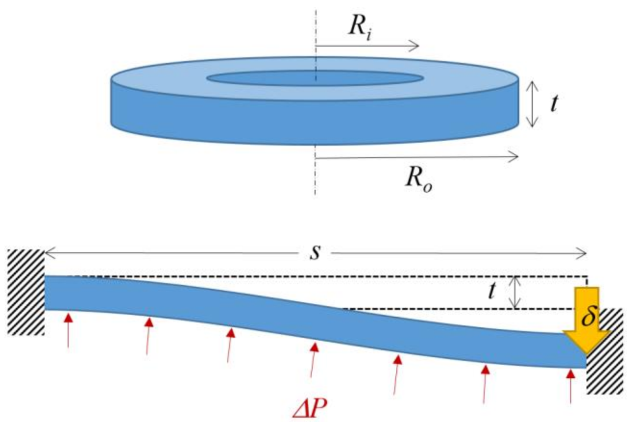


Fig. 8 Geometry for simple bellows analysis

A large displacement δ will require a large number of convolutions N to keep the stress low. This may result in a low fundamental axial resonance frequency ω_n. If ω_n is not sufficiently higher than the forcing frequency ω, then the dynamic stress may be significantly higher than the static stress. For this bellows, this was definitely the case, so we will want to maximize ω_n.

If we disregard ΔP for simplicity, then we can define the number of washers in terms of stress:

$$N \cong \left(\frac{3E}{1-\nu^2}\right)\left(\frac{t}{s^2}\right)\left(\frac{\delta}{\sigma}\right). \tag{3}$$

The stiffness K per unit circumference C for one washer is

$$\left(\frac{K}{C}\right)_1 \cong \frac{2E\left(\frac{t}{s}\right)^3}{(1-\nu^2)} \tag{4}$$

And the stiffness for N washers for average circumference is

$$K_N \cong \frac{\pi(R_o+R_i)E\left(\frac{t}{s}\right)^3}{N(1-\nu^2)}. \tag{5}$$

The mass of N washers is

$$m = 2N\rho t\pi(R_o^2 - R_i^2) = 2N\rho t\pi s(R_o + R_i). \tag{6}$$

Knowing that the square of the resonance frequency ω_n is proportional to the ratio of stiffness to mass, then we can say

$$\omega_n^2 \propto \left(\frac{E}{1-\nu^2}\right)\left(\frac{t}{s^2}\right)^2\left(\frac{1}{2\rho N^2}\right). \tag{7}$$

If we substitute equation 3 for N into equation 7, we find that

$$\omega_n^2 \propto \frac{\sigma^2(1-\nu^2)}{18E\rho\delta^2}. \tag{8}$$

or simply

$$\omega_n^2 \propto \frac{\sigma^2}{\rho E}. \tag{9}$$

for similar ν and assuming the displacement δ is constant. Maximizing the frequency ratio $S_{max}^2/(\rho E)$ would maximize the axial resonance frequency, where S_{max} is the maximum allowable stress of a given material (e.g. fatigue strength divided by a factor of safety). This frequency ratio was used to screen candidate materials.

The properties of several candidate materials are given in Table 2. Note that even though the titanium alloy has a lower fatigue strength than the other materials, its frequency ratio compares well to the 17-4PH H900 due to its low density.

Table 2 Candidate Materials

	Ti 6Al-4V	17-4 PH H900	AerMet 100	AerMet 310
Ultimate Tensile Strength, MPa (ksi)	965 (140)	1310 (190)	2082 (302.0)	2172 (315.0)
Yield Strength, MPa (ksi)	896 (130)	1172 (170)	1779 (258.0)	1896 (275.0)
Density, kg/m^3 (lb/in^3)	4430 (0.16)	7800 (0.282)	7890 (0.285)	7970 (0.288)
Fatigue Strength 10^7 cycles, R=-1, MPa (ksi)	379 (55)	614 (89)	945 (137)	1034 (150)
Young's Modulus, GPa (Msi)	118 (16.9)	199 (28.5)	192 (27.9)	192 (27.9)
Poisson's Ratio	0.31	0.27	0.305	0.32
Frequency Ratio, m^2/s^2	275	243	590	699

IV. Variable Inner Diameter Design

The first design for this application was a 50-convolution bellows with a wall thickness of 0.5 mm. The static stress met the requirement for maximum allowable stress. However, the fundamental axial resonance frequency was 54 Hz, very close to the operating frequency, and the maximum dynamic stress was too high. This dynamic stress varies with axial location (Fig. 9). This is similar to the way in which the axial stress varies in bar vibration. If a bar is fixed at one end, and statically displaced axially at the other, the deflection along the bar's length is linear, and the stress is constant. However, if the bar is dynamically displaced at one end at its resonance frequency, the deflection along the length is a half sine-wave, and the stress is a half cosine-wave. The maximum dynamic stress is $\pi/2$ times, or about 60% higher than, the static stress (Figs. 10 and 11).

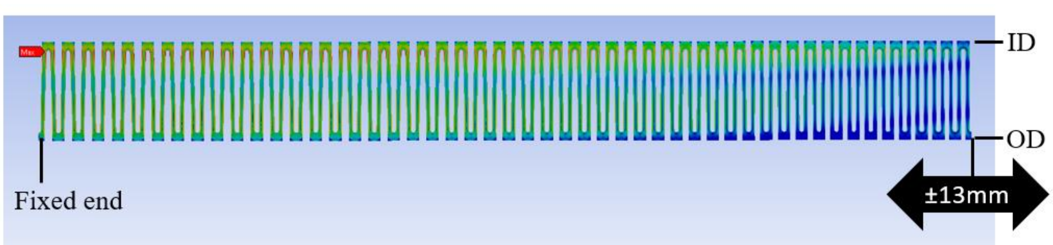


Fig. 9 Bellows cross section, dynamic stress distribution at 55 Hz, high stresses in red, low stresses in blue

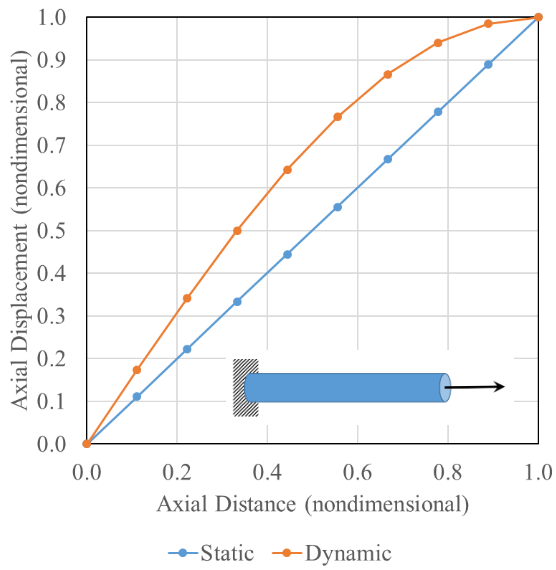


Fig. 10 Bar axial displacement

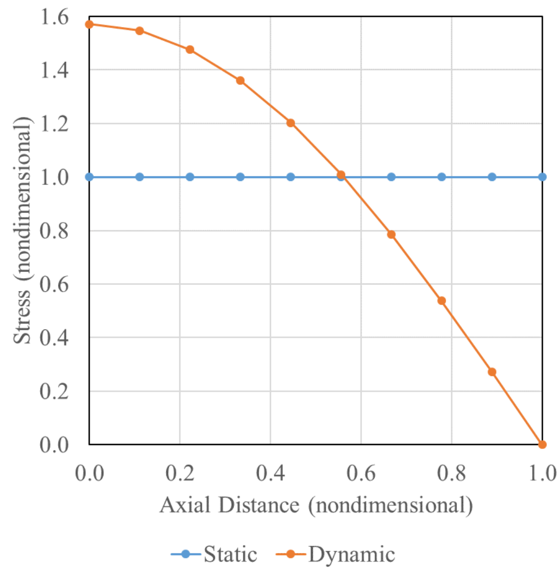


Fig. 11 Bar stress

To distribute the stress more equally among the bellows convolutions, the design was changed so that the inner diameter varied along its axial length. To meet the dynamic stress requirement, the number of convolutions increased to 74. Figure 12 shows the updated 74-convolution tailored bellows with inner diameter values that follow a sinusoidal shape. This stress due to harmonic response at 55 Hz is more evenly distributed along its length. A close-up of the maximum stress area is shown in Fig. 13. Figure 14 shows the maximum dynamic stress vs. frequency for two 74-convolution bellows designs made of AerMet 100 – one with a constant inner diameter and one with a variable inner diameter. The constant ID bellows has a lower static stress, but with a resonance frequency of 37 Hz, its dynamic stress at 55 Hz is too high. The variable ID bellows has significantly lower stress at 55 Hz, and meets the stress requirement.

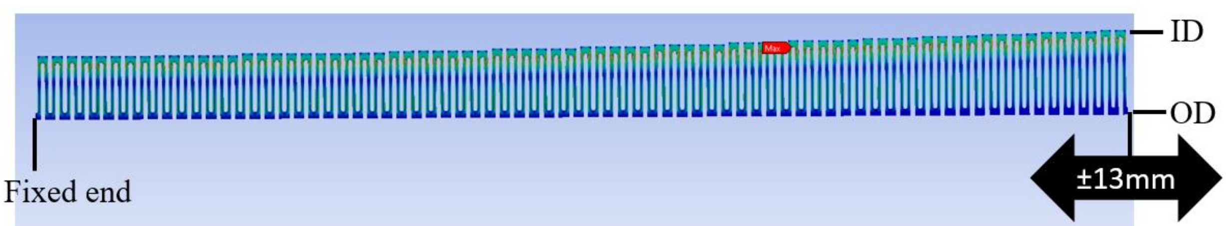


Fig. 12 Variable ID bellows cross section, dynamic stress distribution at 55 Hz, high stresses in red, low stresses in blue

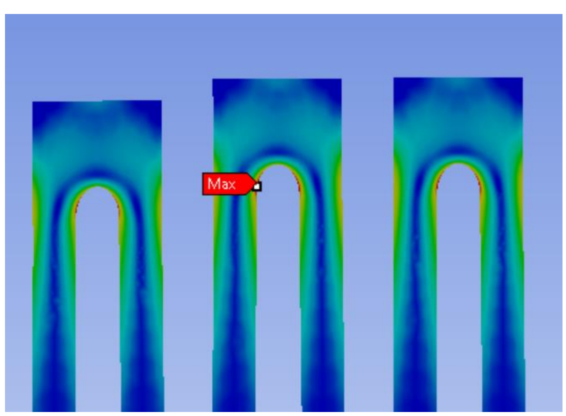


Fig. 13 Detail of maximum stress location on inner diameter slot

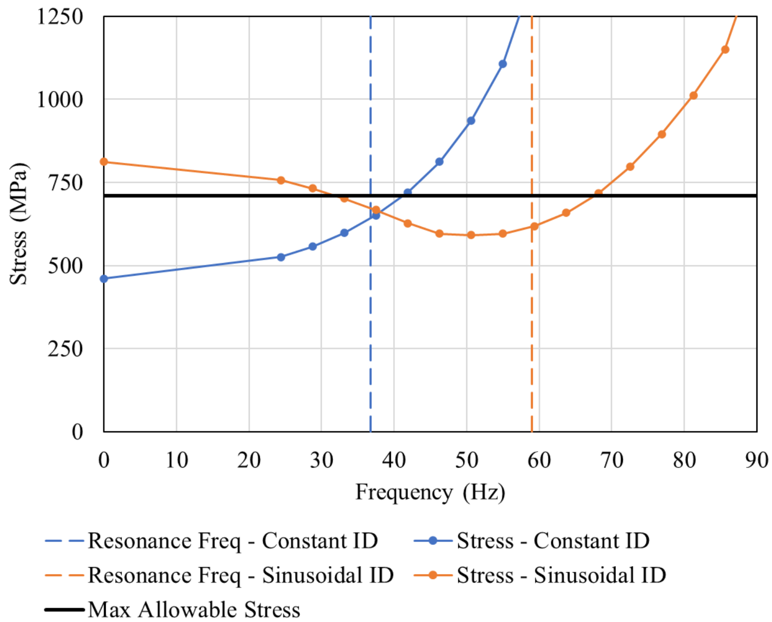


Fig. 14 Maximum stress vs. frequency for constant and sinusoidal ID – AerMet 100

The 74-convolution bellows has a wall thickness of 0.5 mm and a slot width of 0.5 mm, which was the smallest allowable feature for 3D printing. This results in a single convolution axial thickness of 2 mm and a total axial length of 148 mm, well above the original desired axial length of 60 mm. The outer diameter is 44.3 mm to meet the requirement, and the minimum inner diameter is 20.3 mm.

The convolution wall thickness could be smaller or larger, which affects the inner diameter of the bellows, since the ratio of wall thickness *t* to span *s* needs to remain the same. Therefore, there will be an upper limit on the wall thickness that will result in a reasonable inner diameter value. Changing the wall thickness has a minor effect on the total bellows length, so other factors such as machinability, weld precision, or cost can drive that design decision.

V. Test Articles

To determine whether a bellows could withstand the stresses induced during operation, a short bellows was designed for testing including only 10 convolutions (Fig. 7). The bellows samples were fabricated from 17-4PH H900, which was chosen based on availability of this material for manufacturing using two different methods. Note that its fatigue strength is 35% lower than the AerMet 100 specified for the 74-convolution design.

The first test article was made using direct metal laser sintering (3D printing) with a resolution layer thickness of 30 microns. A short bellows was produced using this process, and electropolished to give the surface finish shown in Fig. 15. The inner radius of the outer slot was further polished to increase the fatigue strength at that high-stress location.

A second short bellows is currently being fabricated by producing individual convolutions by machining and wire EDM (Fig. 16). These convolutions will be laser micro-welded at their outer diameters. The inner radius of the outer slot will be polished as well. Some additional welded test articles were made to determine the stiffness and strengths of two different weld configurations at their outer diameters prior to fabrication of the full short bellows. A cross section of a pull test sample is shown in Fig. 17. These pull test samples consist of two unslotted convolutions welded together at their outer diameters, with two different weld fill angles of 45 and 60 degrees (see triangular weld shape in Fig. 6). Both test articles showed no plastic deformation during stiffness testing to a force at the expected yield strength, with constant stiffness, suggesting full weld penetration through the thickness. The next step is to pull test these pieces to ultimate failure. The weld geometry with the highest strength will be used to fabricate the full short bellows for fatigue testing.

Fig. 15 3D printed short bellows

Fig. 16 Individual convolutions prior to outer diameter weld

Fig. 17 Cross section of test article for weld pull test

Fig. 18 Pull test rig

VI. Fatigue Testing

Both the 3D printed and welded short bellows will be fatigue tested, and their results compared. This test is meant to determine whether the material and geometry can survive to 10^7 cycles. The test rig (Fig. 19) has a motor that connects to a crankshaft that produces a fixed axial displacement amplitude to one end of the bellows at 60 Hz. Unlike the cryocooler application, there will be no pressure differential, only displacement. And unlike the 74-convolution bellows, this bellows has a resonance frequency of 274 Hz, well above the driving frequency of 60 Hz, so the dynamic stress will not be a significant factor.

Table 3 shows the fatigue test plan. Since testing occurs at constant displacement amplitude, a crack or failure might not be easily detected while the test is running, so visual inspections and stiffness measurements at specified intervals will be vital to determining crack initiation or failure. Before testing, each bellows will be inspected, and the baseline stiffness will be measured in the pull test rig. Then at each testing interval, they will be inspected and stiffness measured and compared to the baseline value.

The full displacement amplitude of ±2.5 mm was chosen based on the maximum allowable stress at the inner radius of the outer slot. This maximum allowable stress is the fully reversed fatigue strength divided by a factor of safety of 1.33. An initial fatigue test at half that displacement amplitude will be performed first.

Currently, the 3D printed bellows is in the process of being tested. The initial inspection and stiffness measurements have been done, and 2×10^6 cycles have been successfully completed for the half displacement amplitude of ±1.25 mm.

If either of the two short bellows survives the fatigue test, then the full 74-convolution bellows with variable inner diameter can be fabricated and tested, and the effects of dynamic stress can be investigated.

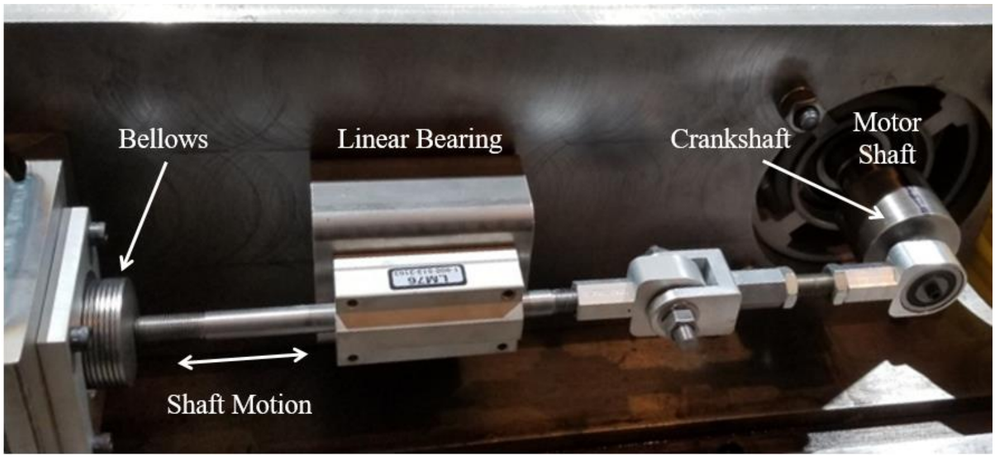


Fig. 19 – Fatigue tester with short bellows prototype

Table 3 Fatigue test plan

Manufacturing technique	- 3D printed - Machined, wire EDM slots, and welded outer diameters
Displacement amplitudes	- 1.25 mm (half range) - 2.5 mm (full range)
At each interval	- Visual inspection - Stiffness measurement
Inspection and measurement intervals	$2x10^4$ cycles (5.6 minutes at 60 Hz), $1x10^5$ cycles (28 total minutes), $5x10^5$ cycles (2.3 total hours), $1x10^6$ cycles (4.6 total hours), and every 10^6 cycles through $1x10^7$ cycles (46 total hours)

VII. Discussion

The dynamic bellows was difficult to design due to the large displacement and pressure differential. This resulted in a bellows that is much longer than the desired 60 mm axial length. Additionally, the axial resonance of the bellows resulted in modifying the inner diameter along the axial length to distribute the maximum stress along the entire bellows. A final design consisting of 74 convolutions made of AerMet 100 was determined to meet the requirements based on finite element analysis, while maintaining a maximum stress below the allowable stress.

To test the fatigue properties of the design, short 10-convolution bellows made of 17-4PH are being fabricated using two different techniques – direct metal laser sintering, and edge welded convolutions with wire EDM slots. The welded test pieces had successful stiffness tests conducted, showing no plastic deformation through the displacement at which yield is expected to occur. Next, both bellows will be tested for fatigue. If the bellows survive the fatigue test to 10^7 cycles at the full displacement, then the full 74-convolution bellows fabrication and testing can proceed. That test article would further show whether the dynamic stress becomes a concern for fatigue life, and whether it can be used as a replacement for a piston in a cryocooler.

The requirements for this bellows are based on the cryocooler design for the NASA HEMM motor. This particular motor has a very large displacement and pressure differential in a relatively small diameter, at a given frequency of near 60 Hz. Work is currently ongoing to optimize cryocooler designs for future motors, including investigating higher-frequency, lower displacement amplitude options. Preliminary investigation of the bellows design with these other requirements show that the bellows may be a better option for those cryocooler concepts.

VIII. Acknowledgments

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